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Dated 6 August 2002

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P01/7700 0.00-0215249.4

Request for grant of a patent

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1. Your reference — DP-308447

2. Patent application number
(The Patent Office will fill in this part)
02 JUL 2002 — 0215249.4

3. Full name, address and postcode of the or of each patent applicant *(underline all surnames)*

DELPHI TECHNOLOGIES, Inc.
P.O. Box 5052
Troy
Michigan 48007
United States of America

Patents ADP number *(if you know it)* 07588332000 1

If the applicant is a corporate body, give the country/state of its incorporation — Delaware, USA

4. Title of the invention

AIR CONDITIONING SYSTEM

5. Name of your agent *(if you have one)* — DENTON, MICHAEL JOHN

"Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)*

Delphi Automotive Systems
31 Carlton Hill
London. NW8 0JX.

Patents ADP number *(if you know it)* 06818777002

6. If you are declaring priority from one or more earlier patent applications, give the country and the date of filing of the or of each of these earlier applications and *(if you know it)* the or each application number

Country	Priority application number *(if you know it)*	Date of filing *(day/month/year)*

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Number of earlier application	Date of filing *(day/month/year)*

8. Is a statement of inventorship and of right to grant of a patent required in support of this request? *(Answer 'Yes' if:*
a) any applicant named in part 3 is not an inventor, or
b) there is an inventor who is not named as an applicant, or
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YES

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Description	8
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Request for substantive examination *(Patents Form 10/77)*	
Any other documents *(please specify)*	

11. I/We request the grant of a patent on the basis of this application.

Signature M J DENTON Date 02/07/2002

12. Name and daytime telephone number of person to contact in the United Kingdom

DP-308447

AIR CONDITIONING SYSTEM

Technical Field

The present invention relates to an air conditioning system having carbon dioxide as the refrigerant, particularly for use in a motor vehicle.

Background of the Invention

As a result of the need to reduce the energy consumption arising from use of automotive air conditioning systems, electronically controlled compressors are being increasingly applied. This permits external control which can be used to advantage in a number of different ways. The most valuable in energy efficiency terms is the management of the evaporation temperature to reduce the necessary amount of reheat to a minimum.

Automotive air conditioning systems with carbon dioxide as refrigerant usually have an electronically controlled compressor, but also require an extra degree of flexibility in the form of an electronically controlled expansion valve. The result of having two control elements means that different combinations of settings of the two devices can yield the same cooling performance. However these different setting combinations will have different energy efficiencies and the challenge for the control system is to control to the condition which yields the highest energy efficiency.

The standard solution to this problem described in the literature is based on the recognition that for any given operating point the required head pressure for optimum efficiency is a simple function of the refrigerant temperature at gas cooler outlet. The control problem reduces to one of establishing the relationship so that for any measured gas cooler outlet temperature the system knows what head pressure to control to. Refinements of the approach have suggested that the optimum head pressure can also be a lesser function of refrigerant pressure in the evaporator.

These conclusions are based on making several simplifying assumptions about the behaviour of the system that may or may not be valid in a

particular case but are certainly not valid in principle over a wide operating range.

The closest known prior art is WO 00/06821 (Suzuki).

Summary of the Invention

The general assumption in the literature is incorrect for two reasons. Firstly the optimum being identified is usually the optimum COP for fixed capacity compressor when it should be for fixed system cooling capacity. This point is recognised in the Suzuki publication where it is shown that the COP optima do not lie in a simple pattern that lend themselves to being identified as a simple function of head pressure and evaporator pressure. However Suzuki makes the mistake of defining an operating band for all the COP optima and claiming that one only needs to control within the band for optimum efficiency. Suzuki does not recognise that each operating point optimum has its own tolerance band and that to lie within the overall band does not mean that optimum COP will be achieved in any particular case. Experimental data indicates that the Suzuki approach will not work.

The basis of the present invention is the improved understanding of the system behaviour and the two possible approaches to achieving system control for optimum efficiency that arise out of this understanding.

The first is an extension of the current standard method of using system performance data in an algorithm to predict the necessary head pressure for optimum efficiency. The algorithm and relevant parameters may result in a more complex solution than the current solution using gas cooler outlet temperature and evaporator pressure, and may need to involve any of the other system operating conditions.

The second proposed solution to the shortcomings identified is to measure or derive values of the relevant parameters that can identify compressor work and use standard Optimum Control techniques to identify the appropriate operating point at which efficiency is a maximum. The most direct approach is to measure compressor torque and thus derive compressor work rate. (The

implementation of torque sensors on compressors is under consideration for other reasons). Optimum Control techniques are used to maintain that value at a minimum at the same time as the system is controlled to meet the required cooling duty. Alternately compressor work can be derived from pressure and flow rate values.

By way of further explanation, real systems differ from the ideal in that as head pressure is changed, by what ever means, there are associated changes in just about every other operating parameter within the refrigerant loop and significant deviation from the idealisation is seen. For example, as the expansion valve is closed to increase head pressure, mass flow rate changes, heat exchanger effectiveness and thus fluid outlet conditions adjust and isentropic efficiency of the compressor changes as pressure ratio changes. Despite the significant deviations from the idealisation, real systems do tend to show a similar behaviour with clearly recognisable maxima in system efficiency as head pressure is varied systematically for a given operating condition.

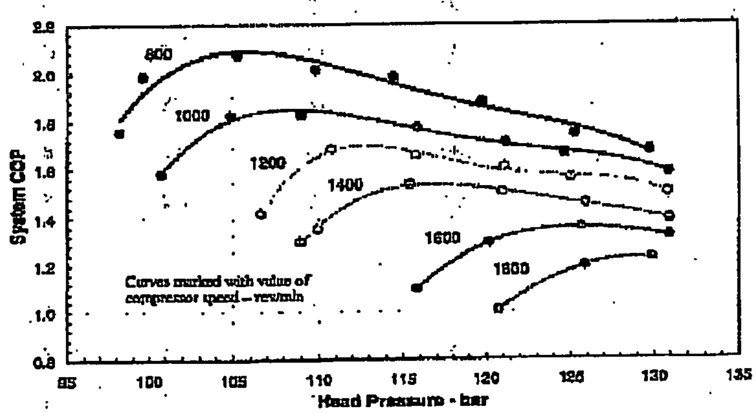

Figure 1. Curves of COP against head pressure for different compressor speeds

An example is given in Figure 1 which shows results taken from a system test stand in which the compressor is electrically driven. In this case all the curves are for fixed air side operating conditions on the evaporator and gas cooler and the different curves are for different compressor speeds. Clear maxima can be seen in the curves as head pressure is varied by adjustment of

the expansion valve and efficiency is seen to fall off rapidly as head pressure is reduced to lower values. This kind of experimental data has been used to derive calibration curves for control to optimum COP. In some cases it has been found necessary to make the control curve a function of evaporator pressure as well as gas cooler outlet temperature.

At this stage it is necessary to consider the detail of how a system is to be controlled. With a fixed displacement compressor at fixed speed (or a speed not subject to external control as for example in an automotive application) only the expansion device can be adjusted to modify performance. As this is adjusted to control the head pressure for optimum COP then the cooling capacity is also changed. This control method works as long as the capacity of the machine at the optimum efficiency more than meets the requirement. The compressor is simply cycled to match performance to load. If the capacity is insufficient at the optimum efficiency condition, further capacity can be extracted by increasing the head pressure. This can increase capacity to meet the need, albeit at a reduced level of efficiency. This understanding of system behaviour can be simply exploited for the efficient operation of an air conditioning system as long as the basic system control method depends on compressor cycling.

The above relatively simple control strategy has to be reconsidered if the system has means for adjusting the capacity of the compressor, either by use of an externally controllable variable compressor or by the use of electric drive with a variable speed motor. The maxima seen in figure 1 no longer, necessarily, represent the targets to be aimed for by the control system. With two independent control means – the expansion valve and the compressor capacity – to be played with, there exist different combinations of the two control settings that yield the same cooling capacity. The control challenge for any operating condition is to identify that combination which delivers the required capacity at the highest efficiency. The point is, that one is now looking for the head pressure at which:-

$$\left(\frac{\partial \varepsilon}{\partial p_h}\right)_Q = 0$$

rather than the point given by curves of the type shown in figure 5:-

$$\left(\frac{\partial \varepsilon}{\partial p_h}\right)_{Displ} = 0$$

where ε is system efficiency, p_h is head pressure, Q is cooling capacity and *Displ* is compressor displacement in m^3/s.

The extent to which the two parameters differ from each other depends upon the shape of the displacement/head pressure/capacity surface. Very little mention of this fact has been made in the open literature. The above mentioned Suzuki publication is the only reference found to the need to identify the head pressures for optimum COP from iso-capacity curves.

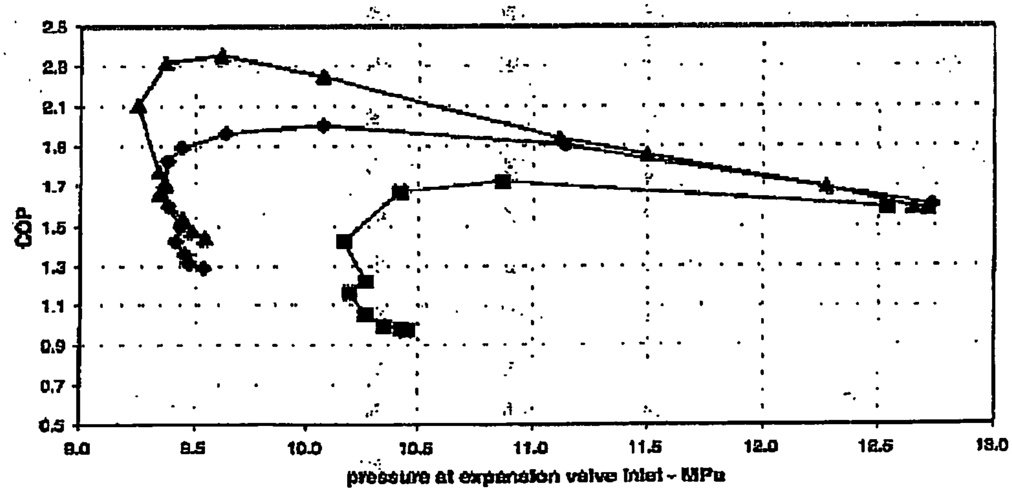

Figure 2. COP plotted against pressure at expansion valve inlet for constant cooling capacity.

A system test stand was first used to develop the curves shown in figure 1. Initial operating conditions were set with fixed air flows and

conditions at evaporator and gas cooler. Then at a fixed compressor speed the expansion device was closed until the head pressure reached a value around 13 MPa and the system allowed to stabilise when data were recorded. The expansion valve was opened in steps, with readings taken after stabilisation at each step until the system no longer delivered any cooling. The whole process was repeated for a number of different compressor speeds. As already stated, the pattern of curves shown in figure 1 is consistent with many similar results from the literature.

After recognising that the location of the maxima given by such curves do not necessarily represent the appropriate target for running a variable compressor system at optimum efficiency, a second series of tests were performed in which, as before an initial head pressure value of around 13 MPa was set. At this stage the capacity of the system was noted before opening the expansion valve by a small amount. The speed of the compressor was then adjusted until the capacity of the system given by the first reading was again reached and the data was recorded. This process was repeated, each time the expansion valve was opened further by a small amount and the compressor speed adjusted until the system gave the same initial capacity value. As the expansion valve is opened the head pressure is reduced but at the same time the cooling capacity drops so compressor speed has to be increased to compensate. In this way an iso-capacity curve was developed. Curves for different capacities were given by starting each series at 13 MPa but with different initial compressor speeds. A family of such curves is shown in figure 2. If they are compared with the curves from figure 1, it can be seen that as the head pressure reduces the COP climbs to a maximum as before. However, as the COP starts to fall off with further opening of the expansion valve, a stage is reached at which compressor speed has to be increased again to maintain the cooling capacity and head pressure starts to climb again. The result is the hook shaped curves shown in the figure.

To identify appropriate control values together with the requisite accuracy, one looks again at the relationships between refrigerant temperature at

gas cooler outlet and the head pressure at which optimum COP is seen. The relevant relationships for the data set presented in figure 2 are shown in figure 3.

The first point to note on the iso-capacity curves is that they are not necessarily vertical lines, but that any change in head pressure along a curve is associated with a change in refrigerant temperature at gas cooler outlet. The challenge for the control system is to find the location of the point of optimum efficiency along the iso-capacity curve within an acceptable tolerance band.

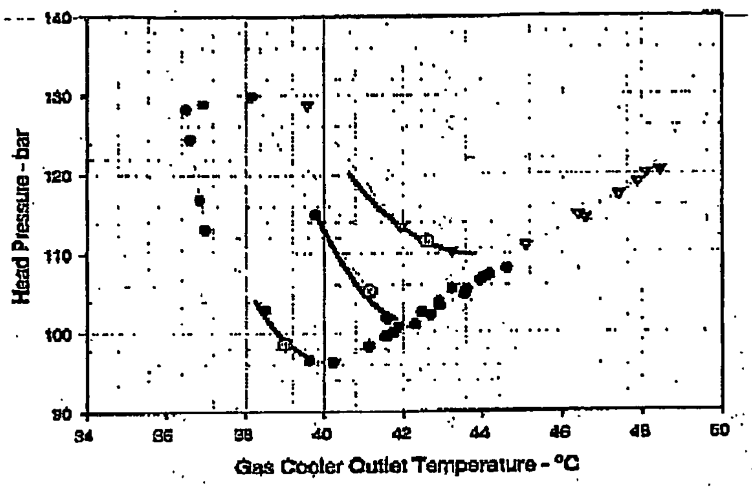

Figure 3. Iso-capacity curves of head pressure plotted against refrigerant temperature at gas cooler outlet

The second point of note is that the acceptable tolerance band remains fairly wide for these particular operating conditions.

Iso-capacity curves have been identified for a wide range of operating conditions and from the curves, acceptable tolerance bands have been identified. The results are shown in figure 4 where each curve represents an operating condition, the each point represents the corresponding condition for optimum efficiency and the curve defines the part of the isocapacity curve that lies within 5% of the optimum COP for that curve.

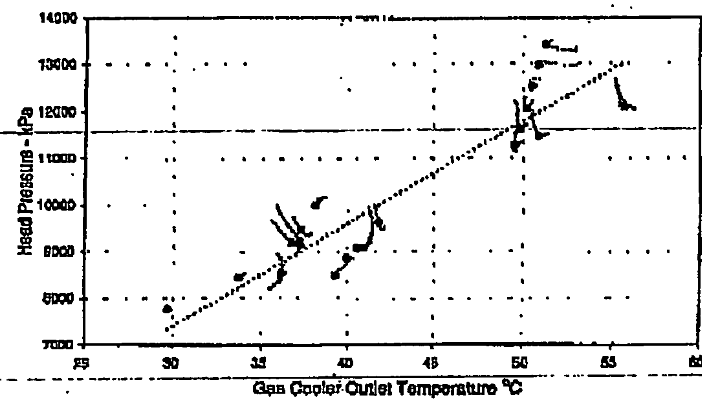

Figure 4. **Isocapacity curves showing optimum efficiency and 5% tolerance band**

The optimum COP points show considerable scatter as has been demonstrated by Suzuki. However it is also clear that no single band can be defined that ensures optimum efficiency will be achieved or even approached in all cases.

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